10/576523
IAP15 Rec'd PCT/PTO 20 APR 2006

Heat exchanger

The invention relates to a heat exchanger, in particular for a motor vehicle, as described in the preamble of claim 1.

Extensive measures, such as for example increased supercharging, more accurate influencing of the combustion conditions, are required to satisfy the increase in demands imposed on modern engines with regard to reduction of emissions and fuel consumption. This also leads to more arduous conditions of use for motor vehicle heat exchangers, specifically higher gas and coolant pressures, increased temperatures and greater volumetric throughputs. At the same time, the demands imposed on power density and service life are also increasing. In some cases, therefore, new cooling concepts are required. For example, in the case of charge-air coolers, the air/air coolers which have customarily been used are at least in part being replaced by air/liquid coolers in order to achieve the required performance and power densities required on account of the high engine supercharging. In the case of exhaust-gas heat exchangers, ever higher exhaust-gas recirculation rates are required, involving likewise evermore arduous operating conditions in terms of pressures, temperatures and power densities. Therefore, ever higher mechanical stresses are encountered in modern heat exchangers, in particular with regard to pressure and oscillations.

High temperature differences between the primary medium, which is to be cooled and is generally in gas form, and the cooling secondary medium, which is generally in liquid form, lead to different levels of component heating on the primary side and secondary side. In the case of exhaust-gas heat exchangers, the

temperature difference may amount to over 700 K, and in the case of charge-air coolers up to 300 K. On account of different thermal longitudinal expansions between the primary and secondary sides, considerable thermal stresses are produced. In the event of rapid changes in operating state, these thermal stresses may also be exacerbated by uneven temperature distributions (thermal shocks).

Moreover, on account of higher heat exchanger power densities, the risk of the coolant boiling rises, which can lead to increased power losses and shorter service lives.

Finally, the processes and materials used are subject to considerable restrictions on account of the occurrence of highly corrosive medium, e.g. condensate from the exhaust gas in the case of exhaust-gas heat exchangers, which given the ever increasing demands imposed on power density leads to ever greater problems in providing a long-term technical solution for balancing sufficient resistance of the flow channels to internal and external pressures with sufficient resistance to induced oscillations and thermal stresses, while avoiding boiling.

It is an object of the invention to provide an improved heat exchanger.

This object is achieved by the heat exchanger having the features of claim 1. Advantageous configurations form the subject matter of the subclaims.

The invention provides a heat exchanger having a housing and at least one tube arranged in the housing, structures being provided between the tubes and the housing and/or between the individual tubes. The

primary medium flows through the tubes. The secondary medium is passed within the spaces between the individual tubes and/or between the tubes and the housing, in which the structures are also arranged. The structures increase the strength by providing a stiffening action with respect to internal and external pressures acting on the tubes. Moreover, the coupling between the tubes and the housing brings about continuous compensation for the thermal stresses between primary and secondary sides over the entire length of the cooler, so that the stresses at the ends of the tubes are considerably reduced. The structures are also used for fluid diverting and distribution within the heat exchanger. Furthermore, the finned metal plates allow better heat transfer, with the result that thermal stresses can be reduced by the improved heat transfer. The increased heat transfer surface area leads to better cooling of the tubes, and boiling can be avoided. Overall, therefore, the result is a considerable increase in the power density of the heat exchanger compared to conventional heat exchangers without structures. As the structures, it is preferable for sheet-metal structures in the form of separate tubes, finned metal plates, studded metal plates or the like to be introduced. The heat exchanger may in particular be an exhaust-gas heat exchanger or charge-air cooler, but may also be another form of heat exchanger, for example another gas-liquid heat exchanger, in which hot gas flows through the heat exchanger (cooler) in tubes in order to be cooled, a liquid-gas heat exchanger, in which cold gas flows through the heat exchanger (heater) in tubes in order to be heated, or a liquid-liquid heat exchanger. As an alternative to using sheet-metal structures, it is also possible for the tubes and/or the housing to be correspondingly designed with structures, i.e. in particular the tube surface may be of fin-like and/or

stud-like design. The structures preferably have a height of from 1 mm to 5 mm, preferably 1 mm to 3 mm, particularly preferably 1.5 mm. The pitch L of the structures is preferably 0.1 to 6 times, particularly preferably 0.5 to 4 times, the structure height h. The transverse pitch Q is preferably 0.15 to 8 times, particularly preferably 0.5 to 5 times, the structure height h. The ratio of passage height between the tubes and passage height within the tube in the region of structures is preferably from 0.1 to 1, preferably from 0.2 to 0.7. The hydraulic diameter between the tubes in the region with structures is preferably from 0.5 mm to 10 mm, for preference from 1 mm to 5 mm.

It is preferable for the structures to be fixedly joined to the housing and/or the tubes, in particular by soldering. In this case, in particular a fixed connection over a large part of the length of the heat exchanger, with or without interruptions, for example to improve distribution of coolant, is provided. The fixed connection very efficiently increases the resistance to external pressure (excess pressure on the secondary side), since the structures provided high rods which prevent the tube from collapsing. Furthermore, oscillations in the relatively labile tubes of conventional heat exchangers are damped by the structures, and the thermal stresses are very efficiently equalized. Furthermore, the fixed connection assists with the heat transfer from the tubes to the structures, resulting in better cooling of the tubes. Moreover, the number of tubes can be reduced by an improved heat transfer, so that the production costs can be lowered.

The tubes are preferably at least in part formed by flat tubes. Flat tubes in this context have a significantly better thermodynamic performance than

round tubes but have a lower ability to withstand pressure, and consequently measures for increasing the ability to withstand pressure are required for flat tubes, such as in accordance with the invention a supporting structure on the outer side of the tubes. The flat tubes in particular have an approximately rectangular cross section with rounded corners. Furthermore, it is possible to provide single-piece rectangular tubes. These may have a longitudinal seam which may be welded, for example laser-welded, friction-welded, induction-welded, or soldered. The rectangular tubes may also be constructed from shells which are welded or soldered together. The tubes may also have any other desired form, for example oval, and/or may have lateral tabs which are soldered and/or welded. Furthermore, to compensate for tolerances between housing and tubes and the structures arranged between them, the tubes can be of slightly convex design. It is also possible for turbulators (winglets) to be provided in and/or on the tubes. The tube surface (inner and/or outer) may also be structured so as to generate turbulence.

It is preferable for the structures at least in part to have an inhomogeneous construction, with the result that coolant can be supplied in targeted fashion to critical regions, so that overheating or boiling can be avoided. A correspondingly increased supply of coolant can also be achieved by the partial emission of structures. The pressure loss in the heat exchanger and the transverse distribution of the coolant in the heat exchanger can be optimized by these measures. The regions with inhomogeneous structures are preferably in the inlet and/or outlet region of the fluid. They are used in particular for flow diversion and to minimize the pressure loss.

The stability of the structures can be increased by at least partial toothing, and furthermore the flow paths of the coolant can thereby be optimized.

To simplify the structure of the heat exchanger, the housing is preferably formed in two or more parts, in particular as a U-shaped shell with a cover, in which case a water chamber can be integrated in the cover. In principle, however, a single-part construction, for example with an integrally formed water chamber, is also possible.

Structures can also be provided in the tubes themselves, in which case all the abovementioned structures which may be provided between the tubes can also be integrated in the tubes. The structures are preferably formed by finned metal plates or studded metal plates which are joined to the tube for example by welding, soldering or clamping. The structures preferably have a height of from 1 mm to 5 mm, preferably 1 mm to 3 mm, particularly preferably 1.5 mm. The pitch L of the structures is preferably 0.5 to 6 times the structure height h. The transverse pitch Q is preferably 0.5 to 8 times the structure height h. The hydraulic diameter in the tube in the region having structures is preferably from 0.5 mm to 10 mm, for preference 1 mm to 5 mm.

The text which follows provides a more detailed explanation of the invention on the basis of an exemplary embodiment and with reference to the drawing, in which:

Fig. 1 shows a section through an exhaust-gas heat exchanger,

Fig. 2 shows a perspective view of the heat exchanger from fig. 1,

Fig. 3 shows a diagrammatic perspective view of a finned metal plate,

Fig. 4 shows a diagrammatic perspective view of a finned metal plate in accordance with a variant, and

Fig. 5a-d show a number of variants of inlet regions.

An exhaust-gas heat exchanger 1 has a two-part housing 2 and a plurality of tubes 3 arranged in this housing 2. Finned metal plates 4 are provided between the individual tubes 3 and between the housing 2 and the tubes 3 as structures, these finned metal plates 4 in accordance with the present exemplary embodiment being toothed, as illustrated in fig. 3 and described in more detail below. The tubes 3 are in the present case flat tubes.

The exhaust gas which is to be cooled and comes from the engine (gaseous primary medium) is passed through the individual tubes 3; the direction of flow is indicated by two solid arrows in fig. 2. The housing 2 in which the tubes 3 are arranged comprises a U-shaped first housing part 2′ and a housing cover 2′′ which is fitted onto the first housing part 2′ from above. Two coolant connection pieces 5 are provided in the housing cover 2′′ as inlet and outlet for the coolant (liquid secondary medium), the direction of flow of the coolant in co-current operation being represented by dashed arrows in fig. 2. Flow in counter-current mode is also possible, for which purpose the direction of flow is reversed. Since the coolant is passed through the

housing 2 and around the tubes 3, the finned metal plates 4 are arranged on the coolant side.

The finned metal plates 4 formed with straight toothing make it easy for the coolant to pass through in the direction of the arrow represented by a solid line in fig. 3 and more difficult for the coolant to pass through in the direction indicated by a dashed arrow in fig. 3. The flow can be influenced by changing the longitudinal pitch L and transverse pitch Q as well as the fin height h. In addition to a straight toothing, oblique toothing is also possible. Given a suitable configuration of the individual finned metal plates 4, these plates can also assist the passage of coolant in targeted fashion at particularly critical locations, for which purpose the finned metal plates 4 are inhomogeneous at least in regions.

Fig. 4 illustrates a simple variant of a finned metal plate with a fin running in a straight direction which has a longitudinal pitch L of 2.4 mm and a fin or structure height h of 1.5 mm. The finned metal plate may also be bent from a perforated metal plate, so that the individual corrugation flanks are permeable on account of the perforations.

According to a variant which is not illustrated in the drawing, a corresponding construction is used for a charge-air cooler.

Fig. 5a-d show various inhomogeneous regions of the structures which form the finned metal plates 4. These effect better distribution of the fluid as it flows in. According to the first variant, illustrated in fig. 5a, transverse distribution passages are provided by deformation or stamping. According to the variants illustrated in fig. 5b and 5c, the finned metal plates

4 have been partially cut away. Fig. 5d shows a variant with a special distributor structure formed on the finned metal plate 4. An inhomogeneous region corresponding to fig. 5a to 5d may also be provided on the outflow side.